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MODEL MODIFICATION OF WIRA CENTER MEMBER BAR

F.R.M. Romlay & M.S.M. Sani
Faculty of Mechanical Engineering
Kolej Universiti Kejuruteraan & Teknologi Malaysia
(KUKTEM), Karung Berkunci 12
25000 Kuantan Pahang
tel. no:609-5492209 fax no:609-5492244
e-mail: fadhlur@kuktem.edu.my, mshahrir@kuktem.edu.my

Abstract
This paper presents the model modification or updating results of WIRA center member bar using finite element method. The loading condition affects the vibration behavior of center member bar, which can make uncomforted or harshness to the passenger. Therefore, experimental modal analysis for center member bar should be carried out to investigate resonant frequency and the mode shapes of the model. Resonant frequency and mode shapes of the structure can be obtained through computer simulation. The model parameters of the structure can be obtained by curve fitting set of impulse response functions (time domain). The time domain functions were changed to the frequency domain functions and compute by the modern Fast Fourier Transform (FFT) based analyzer for deriving operating deflection shapes (ODS). From the analysis, a mode shape result was not suitable for structure frequency. Then, the model modification of the structure was done by using FEMtools™ software to get the simulation resonant frequency was closer to the experiment results.

Keywords: Vibration, Modal Analysis, Resonant Frequency, Mode Shapes, Finite Elements Method

1. Introduction

Resonance happens when the frequency of a vibrating force is equal to a natural (resonant) frequency of the object to which the force is applied. When resonance occurs, a large amount of energy can be transmitted to the object, which results in a large vibration. Structural dynamics is the field concerned with the behavior of structure subject to dynamic excitation. Resonant vibration is characterized in terms of the modes of vibration of a structure. Modes or resonance are inherent properties of structure and can be determined by the material properties like mass, stiffness, damping properties and boundary conditions of the structure. Each resonance can be defined by a natural frequency, modal damping and a mode shape [1]. A structure's modal parameters (resonant frequency, damping, and mode shape) can be estimated from Finite Element Analysis (FEA) or structural dynamics measurements by experimental modal analysis. However the simulation model always not accurate. This paper will present the result from both of the methods. These results were combined to update the FEA model. The updating model was ready for next analysis. The resonant frequency was found to get the mode shape. If excited, modes or resonance can act like "mechanical amplifiers". Modes can cause excessive vibration responses that are orders of magnitude greater than responses due to static loading. So, it is important to know the resonant or natural frequency and update the model parameter to have a safe structure.

2. Theoretical Background

2.1 WIRA Center Member Bar Properties

In this research, a solid beam and a WIRA center member bar were the model for experiment. The beam dimension is 31 x 15 x 61 mm. The weight is 254.1 g, density was 7850 kg/m^3 and Young Modulus is 215GPa. The length of the center member bar is 9870 mm, width is 1135 mm, and high is 328.33 mm. The weight of the bar is 4380 g and the density, ρ is 7850 kg/m^3. The shape is not uniform and the thickness of the model, H is 4 mm. The type of material is steal which the Young Modulus, E is 200Gpa. So, the parameters that will influence the physical and geometry properties need to be defined. The parameter such as Young Modulus, density and thickness of the model will affect the characteristic of the component.

For a car chassis, it is important to have a stable structure. The structure always applied with the cycle dynamic force such as vibration. That's why it is important to have a stable center member bar structure to support the engine weight.

The manual model updating has the limit and difficult to get the similarity of the characteristic [2]. That is why a FEMtools™ program was choosing to build a FE model. With the FEMtools™ program, the experiment and simulation result was compared. The comparison was done until the similarity of mode shape value was reached. It was done by fix the value from experiment. The iteration process was used until the maximum iteration was reach to give the updating mode shape and parameter.

2.2 Modal Analysis

Modal analysis has become reliable and economical method of finding the modes of vibration. Modal analysis is a valuable tool for trouble shooting and verification on FEA models [3]. Experimental modal analysis is routinely performed to determined structural characteristics in the form of modal parameters, resonant frequencies and mode shapes. In other words, modal analysis can be defined as the process of characterizing the dynamics of the structure in terms of its modes of vibration. It turns out that the eigenvalues and eigenvectors of the defined normal mode mathematical model are also parameters that define the resonant frequencies and mode shapes of the modes of vibration of the structure. That is, the eigenvalues of the equations of motion correspond to frequencies at which the structure tends to vibrate with a predominant, well-defined deformation. The amplitude of this wave motion on the structure is specified by the corresponding eigenvector. Each mode of vibration, then, is defined by an eigenvalue (resonant frequency) and corresponding eigenvector (mode shape). In many cases, this information is sufficient for modifying the structural design in order to reduce noise and vibration.

2.3 Dynamic Testing

Dynamic testing can be used for troubleshooting noise and vibration problems in existing mechanical systems. These problems can occur because of errors in the design or construction of the system, or as a failure in some of its components. Not only can testing be used to locate a problem, but it can also be used to evaluate fixes to the problem. Finally dynamic testing can be used to construct a dynamic model for components of a structure, which are too difficult to model analytically. In all the cases mentioned above, the objective of the dynamic testing procedure is to excite and identify the test specimen's modes of vibration. This process involves identifying the eigenvalues and eigenvectors of the equations of motion. These parameters also define the modes of vibration of the structure.

2.4 Spectral Analysis

Probably the most convenient way to analyze a vibration signal is to obtain its frequency content, or frequency spectrum [4]. There are at least two good reasons for this:
a) Excitation forces (especially in rotating equipment), of-ten provide sinusoidal excitation at specific frequencies. These forces are manifested as peaks in a frequency spectrum.
b) Resonances are also manifested as peaks in a frequency spectrum. Sine wave generators were used to artificially excite structures, one frequency at a time. Oscilloscopes were used to look at the signals.

2.5 Accept/Reject

Because impact testing relies, to some degree, on the skill of the one doing the impacting, it should be done with spectrum averaging, using 3 to 5 impacts per measurement. Since one or two of the impacts during the measurement process may be *bad hit*s, an FFT analyzer designed for impact testing should have the ability to accept or reject each impact. An accept/reject capability saves a lot of time during impact testing since you don't have to restart the measurement after each bad.

2.6 Time Domain Model

The actual input forces and responses for a finite number of degrees-of-freedom of the structure were measured. S 0 if a model were constructed from the measurements involving these specific degrees-of-freedom, the model would give an accurate description of the structural dynamics involving those points. This is a different situation than with a finite element model where the degrees-of-freedom and the size and shape of the elements are chosen so as to approximate the dynamics of the structure as closely as possible. The time domain dynamic model exhibits the same form as the finite element model but at least in principle, is an exact model of the structural dynamics if obtained from measurements.

$$M\ddot{x}(t)+C\dot{x}(t)+Kx(t)=f(t) \tag{1}$$

2.7 Lap1ace Domain Model.

We do not directly measure the time domain model, but rather its laplace domain equivalent. In this model the inputs and responses of the structure were represented by their laplace transforms. Time domain derivatives (i.e. velocity and acceleration) do not appear explicitly in the laplace domain model but were accounted for in the transfer functions. The transfer matrix contains transfer functions, which describe the effect of input at each degree-of-freedom (D.O.F.) upon the response at each D.O.F. Because the model is linear, the transformed total motion for any D.O.F. is the sum of each transformed input force multiplied by the transfer function between the input D.O.F. and the response D.O.F.
For example:

$$x_1(s)=h_{11}(s)F_1(s)+h_{12}(s)F_2(s)+...+h_{1n}(s)F_n(s) \tag{2}$$

2.8 Rules of Digital Measurement

There are three key equations that govern the use of the DFT. [2] The first one describes the sampled signal in the time waveform (time domain). The DFT assumes that the sampled time waveform contains N uniformly spaced waveform samples, with an increment of (Δt) seconds between samples. (The most common FFT algorithms restrict N to being a power of 2, although this is not necessary.) The total time period of sampling (also called the *sampling window*), starts at (t=0) and ends at (t=T). Therefore,

$$T= (\Delta t)\ N\ \text{(seconds)} \tag{3}$$

The second describes the sampled spectrum in the Frequency Waveform (frequency domain). The DFT assumes that the digital frequency spectrum contains N/2 uniformly spaced samples of complex valued data, with frequency resolution (Δf) between samples. The frequency spectrum is defined for the *frequency range* (f=0) to (f=F_{max}). Therefore,

$$F_{max} = (\Delta f)\ (N/2)\ \text{(Hertz)} \tag{4}$$

Nyquist Sampling or Shannon's Sampling Theorem says that a frequency spectrum can only contain unique frequencies in a range from (f=0) up a maximum frequency (f= F_{max}) equal to one half the sampling rate of the time domain signal. Therefore,

$$F_{max} = (1/2)\ (1/\Delta t)\ \text{(Hertz)} \tag{5}$$

The three equations above can be used to derive the most fundamental rule of digital spectrum based testing, Δf=(1/T). This equation says that the frequency resolution obtain able in a digital spectrum depends on the time domain sampling window length **(**T), not the sampling rate. Stated differently, to get better frequency resolution, you have to sample over a longer time period.

2.9 Transfer Function Method

This method is faster and easier to perform, and is much cheaper to implement than the normal mode testing method. The major steps of the transfer function method are depicted in Figure 1. It is based upon the use of digital signal processing techniques and the FFT (Fast Fourier Transform) algorithm to measure transfer functions between various points on the structure. In a test situation we do not actually measure the transfer function over the entire S-plane, but rather its values along the jω-axis. These values are known as the frequency response function. Since the transfer function is an "analytic" function, its values throughout the S-plane can be inferred from its values along the jω-axis. More specifically, if we can identify the unknown modal parameters of a transfer function by "curve fitting" the analytical to measured values of the function along the jω-axis, then we can synthesize the function throughout the S-plane [5].

Model parameters were identified by performing curve fitting on this set of transfer function measurements. Figure 2 shows how modal parameters can be obtained from transfer function measurements. The figure shows the imaginary part of each transfer function made between an impact point and the reference point. Modal frequencies correspond to peaks in the imaginary part of the transfer functions. A peak should exist at the same frequency in all measurements, except those measured at "node" points where the modal amplitude is zero. The width of the modal peak is related to the damping of the mode. That is, the wider the peak, the higher the modal damping. The mode shape is obtained by assembling the peak values at the same frequency from all measurements. As shown in Figure 2, as modal frequency increases, the complexity of the mode shape also increases. A hammer excitation was used to have the transfer functions.

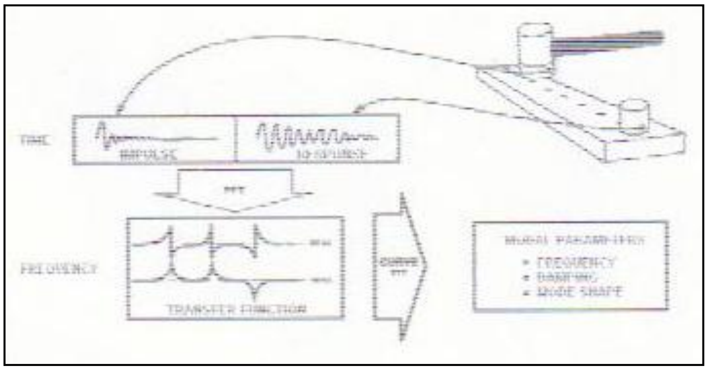


Figure 1: Transfer Function Method

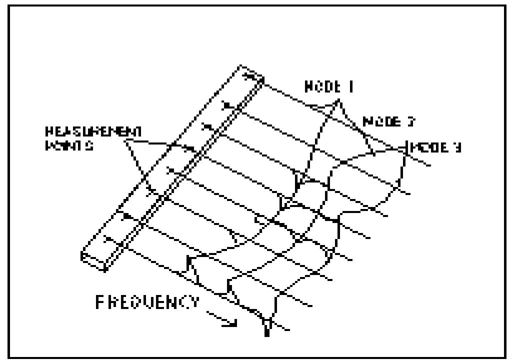


Figure 2: Imaginary Part of Transfer Function

3. Experiment Method

A spectral analysis was done by experiment to obtain its frequency content or frequency spectrum. Both of the components were analysts in a free vibration. Impact testing is the most commonly used method for finding the resonance's of structures. Impact testing requires a minimum of equipment: a hammer with a load cell attached to its head to measure the impact force, an accelerometer fixed to the structure to measure response motion and a 2-channel FFT analyzer for analysis. Of course, different sized hammers are necessary to provide the appropriate impact force to the structure.

The accelerometer was located 150mm from the side at the top of the surface of the beam. For the center member bar, the location is in the Figure 4. The optimum location for the accelerometer was decided by the 'pretest analyses with FEMtools^{TM} software. The displacement of the model was normalized by pretest analyst. Five DOFs (point and direction) were fixed for the beam and 10 DOFs for the center member bar. A set of transfer function was obtained by exciting at each DOF with a load cell, simultaneously measuring the input force and corresponding response motion signals, and then dividing the Fourier transform of the response by the transform of the input. Because the impulse signal exists for such a short period of time, it is important to capture all of it in the sampling window. Sampling data was beginning before the trigger point occurs by the analyzer, which is usually set to a small percentage of the peak value of the impulse. Analog filters were used to limit the frequency content (band limit) the signals. Special analog filters that changed with the frequency of excitation (swept filters) were used to obtain the structural response, one frequency at a time. This response to each excitation frequency is a frequency spectrum. This is called a pre-trigger delay. Once the measurement signals have been digitized and stored in the computer's memory, further processing of the data was performed to reduce the effects of noise and distortion. Statistical estimation algorithms, which use large amounts of measurement data, can also be used to estimate modal parameters with more accuracy. Lastly, the mode shapes can be displayed in animation. All the data was acquisition by PAK^{TM} software. To obtain the resonant frequency, the data was exported to the ME'scope^{TM} software. The time domain model was change to the Fourier transform, which is frequency domain model.

The frequency and damping were recognized by curve fitting. The single mode method was used whenever sufficiently accurate results can be obtained. The simplest single mode methods are shown in Figures 49 and 50. Figure 49 shows that modal frequency is simply taken as the frequency of the peak of the transfer function magnitude. Damping can be obtained by measuring the width of the modal peak at 70.7% of its peak value or by computing the slope of the phase function at resonance. The first method is known as the "half power point" method since 70.7% of the magnitude is the same as 50%, or half, of the magnitude squared.

Finally the residue can be estimated by using the peak value of the imaginary part of transfer function at resonance. This is known as "quadrature picking". At least the lowest 10 peaks rms voltages were choose for each magnitude spectrum. If the frequency resolution is not good, all of the above methods can yield largely incorrect results since they use only one or two data points from the measurement. However, the damping result was not interested because it was ignored in FE analysis. This because it was very difficult to define the accurate value for the model. The value of the damping also very low compare to the mass and stiffness value.

For FEA work, the designed was finished in Autocad™ R14 (Figure 4). It was very hard to design the model in the FEMtools™. The 3D models from Autocad™ R14 was exported to the MSC Nastran™ in the 'Export [*.DXF]' format file. The model was meshed to be a FE model. The triangle-meshed shape was used. The models from MSC Nastran™ was export to the FEMtools™ in the 'FEMLAB*.neu' file. The materials and geometry properties were set in the program such as in the Table 1 and 2.

Table 1: Material Properties for the Beam

#	P	E	NU	γ
1	7.85E-5	2.00E+5	3.30E-1	7.70E-5

Table 2: Material Properties for the Center Member Bar

#	P	E	NU	γ	T
1	7.85E-6	2.00E+5	3.30E-1	7.70E-5	4.0

Overall, 12 elements, 8 nodes and a set of material properties were used for the beam. For the center member bar, 1088 plate elements, 530 nodes and a set of materials properties were used. The thickness of the models was 4 mm.

The mode shape and resonance frequency analyses were done. For the element matrices automatic calculation, the stiffness and mass matrices were choosing. Including the mode quantity and frequency range that was interested, the normal mode analysis was done. From 1400 Hz to 1E30 Hz, 150 mode shapes were asked.

Because of the inaccurate in the modeling, the value of the perimeter can be change by the model updating analysis. Modulus Young, the element thickness and mass density were the parameter that considered. However, in practical, changing the element thickness was the easiest.

4. Results

The simulation results for mode shape and resonant analysis (Figure 5 to 8 and Figure 10 to 12) are different compare to experiment result as in the Table 3 and 4. The design and constant error of manufacturing was the factors that affect the result.

Table 3: Comparison of Mode Shape for Center Member Bar

MOD	FEA (Hz)	EMA (Hz)	Error
1	1883	1891.82	-0.47
2	2165.16	2164.23	0.04
3	2538.11	2542.84	-0.19
4	3421.23	3398.83	0.66

Table 4: Comparison of Mode Shape for Solid Beam

MOD	FEA (Hz)	EMA (Hz)	Error
1	213.67	207.97	2.74
2	1231.51	1078.93	14.14
3	2527.06	2624.43	-3.71

The updating resonant result was present in Table 5 and 6. The changing parameter was present a new resonant frequency and mode shape as shown in Figure 9 and 13.

Table 5: Model Updating of Simulation Mode Shape for Center Member Bar

MOD	FEA (Hz)	EMA (Hz)	Error
1	0.00	0.00	0.0000
2	1891.80	1891.80	-0.0019
3	2164.30	2164.20	0.0033
4	2542.70	2542.80	-0.0045

Table 6: Model Updating of Simulation Mode Shape for Solid Beam.

MOD	FEA (Hz)	EMA (Hz)	Error
1	209.74	207.97	0.85
2	1079.07	1078.93	0.01
3	2604.05	2624.43	-0.78

5. Discussion

5.1 Force & Exponential Windows and Low Pass Filter

Two common time domain windows that were used in impact testing were the force and exponential windows. These windows were applied to the signals after they were sampled, but before the FFT was applied to them. The force window was used to remove noise from the impulse (force) signal. Ideally, an impulse signal was non-zero for a small portion of the sampling window, and zero for the remainder of the window time period. Any non-zero data following the impulse signal in the sampling window was assumed to be measurement noise. The force window pre-serves the samples in the vicinity of the impulse, and zeros all of the other samples in the sampling window.

To reduce leakage in the spectrum of the response, the exponential window was used. If the response decays to zero (or near zero) before the end of the sampling window, then there was no leakage, and the exponential window need not be used. In the response does not decay to zero before the end of the window, then the exponential window must be used to reduce the leakage effects on the response spectrum. The exponential window adds artificial damping to all of the modes of the structure in a known manner. This artificial damping was subtracted from the modal damping estimates. Leakage was removed from its spectrum when the exponential window causes the impulse response to be completely contained within the sampling window.

When it was sampled less than twice the highest frequency in the spectrum of the signal, aliasing of a signal occurred. The parts of the signal at frequencies above the sampling frequency add to the part at lower frequencies. This happen when aliasing occurs and thus giving an incorrect spectrum. The FFT analyzers that have been used guarantee that aliasing will not occur by passing the analog signals through anti-aliasing filters before they were sampled. An anti-aliasing filter band limits (low pass filters) the signal so that it contains no frequencies higher than the sampling frequency. The FFT computes a discretized (sampled) version of the frequency spectrum of a sampled time signal. This discretized, finite length spectrum is called a Discrete Fourier Transform (DFT). Since all filters have a roll off frequency band, the cutoff frequency of the anti-aliasing filters was typically set to *40%* of the sampling frequency. Therefore, *80%* of a DFT frequency band was considered to be alias-free.

5.2 Spectrum Averaging

Spectrum averaging in an option in most modern FFT analyzers. It is done with a spectrum averaging loop. Spectrum averaging is used to remove the effects of extraneous random noise and randomly excited non-linearities. In a spectrum averaging loop, multiple spectral estimates of the same signal are averaged together to yield a final estimate of the spectrum. Different types of averaging can be used, but the most common type (called stable averaging), involves summing all of the estimates together and dividing by the number of estimates. The FFT is a linear, one-to-one and onto transformation. That means that it uniquely transforms the vibration signal from a linear dynamic system into its correct digital spectrum. If a signal contains any additive Gaussian random noise or randomly excited non-linear behavior, these portions of the signal are transformed into spectral components that appear randomly in the spectrum.

5.3 Removing Random Noise & Non-Linearities

By summing together (averaging) multiple spectral estimates of the same signal, the linear spectral components will add up (re-enforce one another), while the random noise and non-linear components will sum toward zero, thus removing them from the resultant average spectrum. In order to remove random noise and non-linearities while retaining the spectral components of the linear dynamics, we must guarantee that the magnitudes and phases of the linear portion of all spectral estimates are the same. This depends on how the data is sampled in each sampling window.

5.4 Free Response Function (FRF)

The Frequency Response Function (FRF) is a fundamental measurement that isolates the inherent dynamic properties of mechanical structures. Experimental modal parameters (resonant frequency, damping, and mode shape) are obtained from a set of FRF measurements. The FRF describes the input-output relationship between two points on a structure as a function of frequency. The FRF is defined as the ratio of the Fourier transform of a motion output (or response) divided by the Fourier transform of the force input that caused the output.

Since both force and motion are vector quantities (they have directions associated with them), each FRF is actually de-fined between input DOF (point and direction), and an output DOF.

5.5 Curve Fitting: Single Mode Methods

Because of their speed and ease of use, single mode methods should be used whenever sufficiently accurate results can be obtained. Damping can be obtained by measuring the width of the modal peak at 70.7% of its peak value or by computing the slope of the phase function at resonance. The first method is known as the "half power point" method since 70.7% of the magnitude is the same as 50%, or half, of the magnitude squared. Finally the residue can be estimated by using the peak value of the imaginary part of transfer function at resonance. This is known as "quadrature picking" or simply the quadrature method. If the measurements are noisy, or if the frequency resolution is not good, all of the above methods can yield largely incorrect results since they use only one or two data points from the measurement. The methods will yield better results in general, since they use more measurement data.

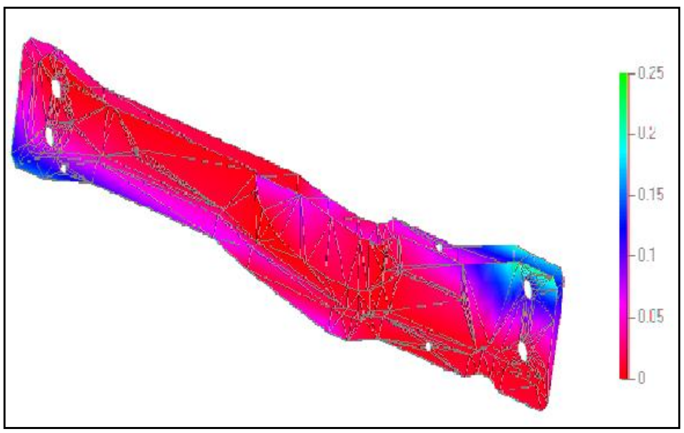


Figure 3: Result of 'Pretest' Analysis

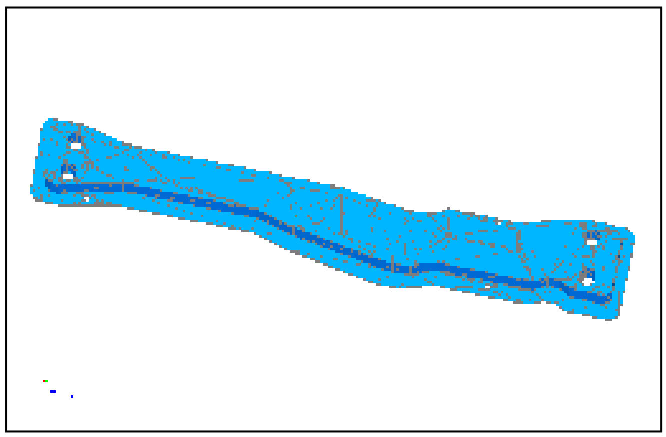


Figure 4: FE Model of Center Member Bar

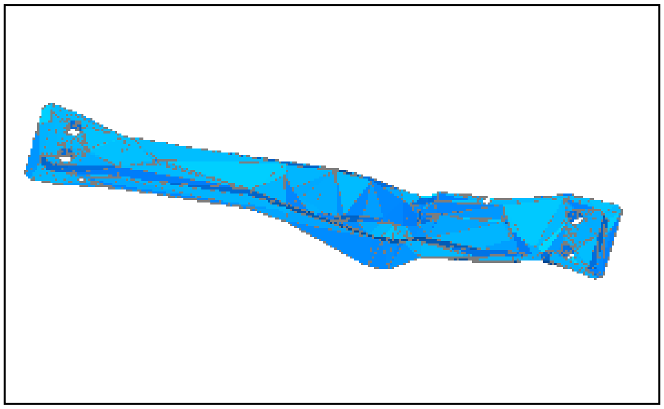


Figure 5: Mode Shape 1

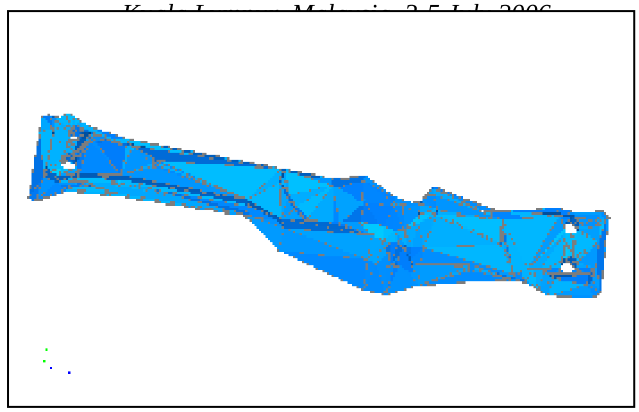


Figure 6: Mode Shape 2

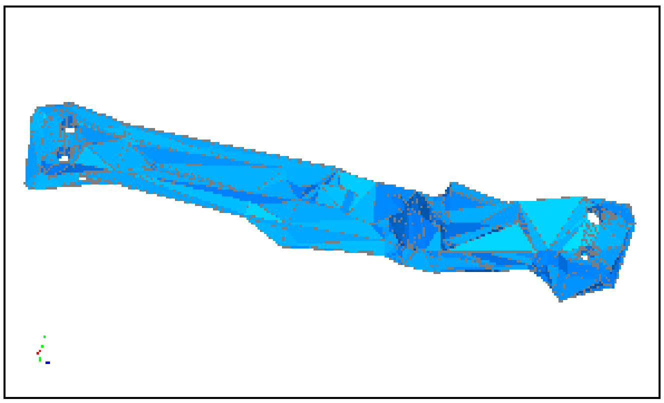


Figure 7: Mode Shape 3

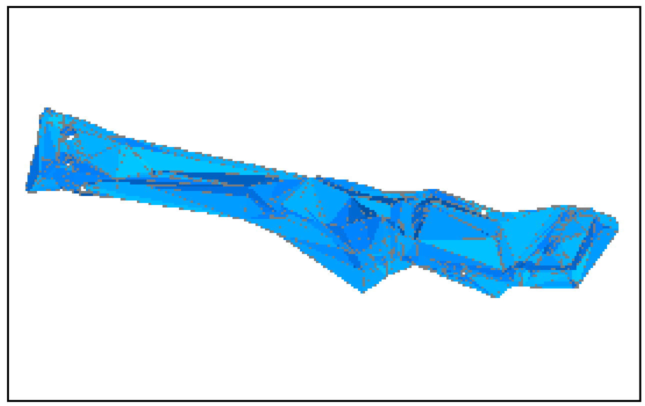
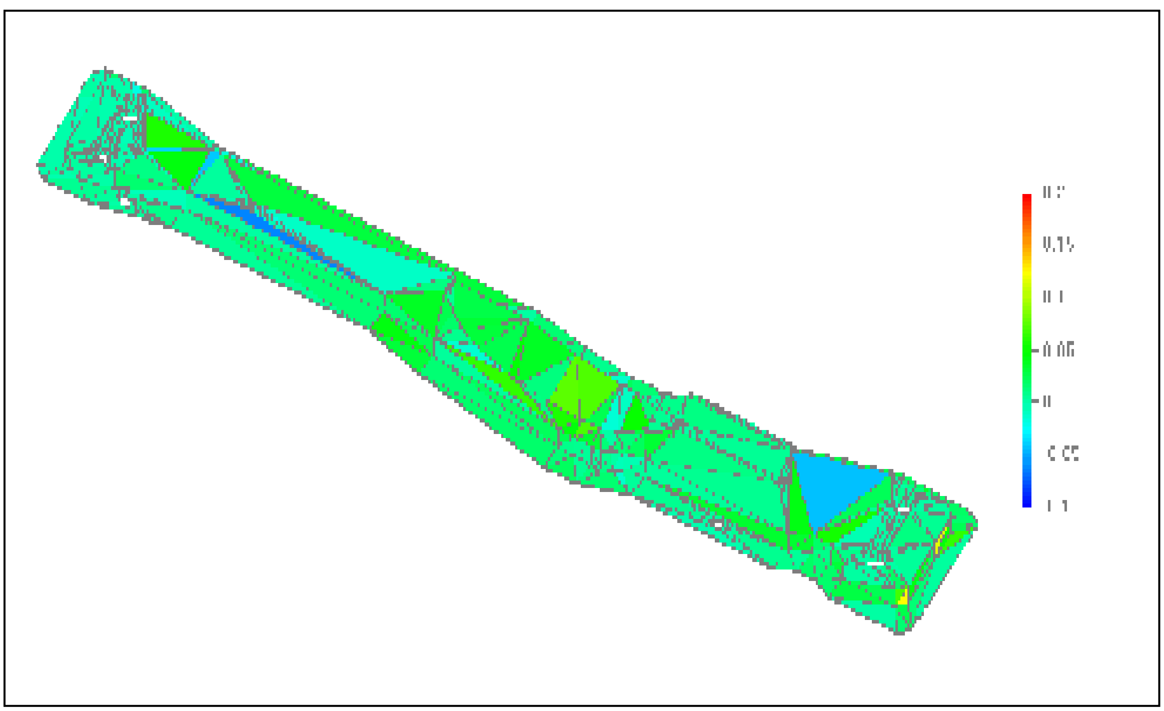
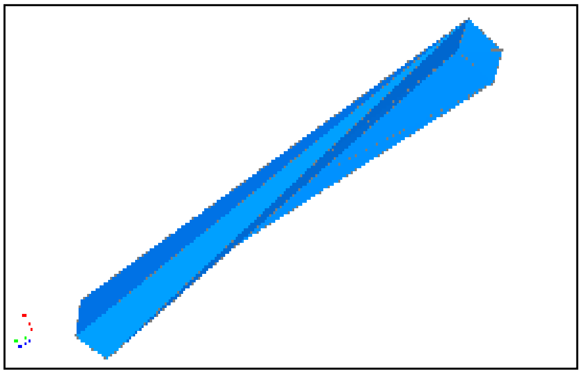


Figure 8 : Mode Shape 4





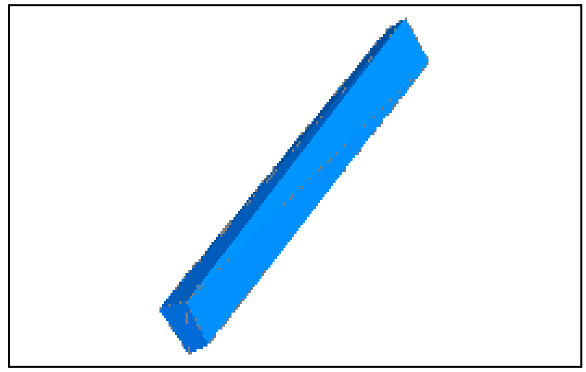
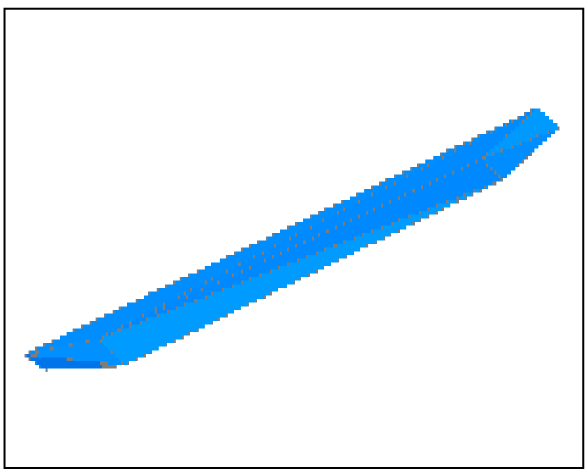


Figure 9: Result of Model Updating for Thickness (H) Analysis

Figure 10 : Mode Shape 1

Figure 11 : Mode Shape 2



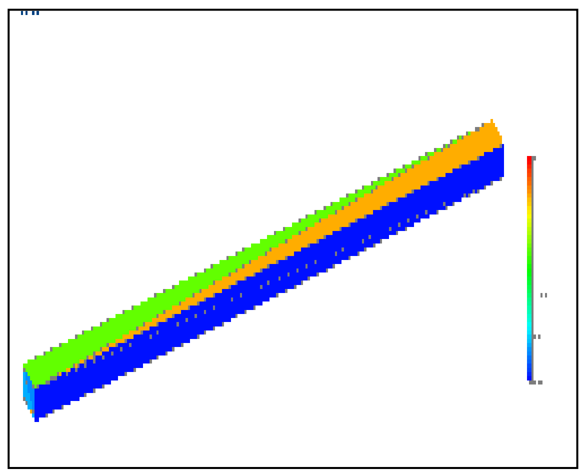


Figure 12 : Mode Shape 3

Figure 13 : Result Of Model For Thickness (H) Analysis

Conclusions

From the analysis, the resonant frequency from the simulation was not accurate. However, the model updating for simulation model can be done if the data from experiment was collected. For the model updating process, the experiment data was been a reference. The model updating was done by FEM, which the selected parameter was modified to have the close value of experiment mode shape frequency. Because of that, the updating simulation model can be used for the next analysis and will give more accurate results.

Acknowledgments

Many of the methods and ideas reviewed here were learned from the software supplier and the literature from Structural Measurement Systems, Inc. and Vibrant Technology, Inc.

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